RESEARCH AND DEVELOPMENT

OF A

HUMAN ANGULAR ACCELERATOR

by

WINSTON C. BOTELER

Summary Report, Phase I

Preliminary Design

April, 1959

Final Report, October, 1963

Project A-425

Engineering Experiment Station
Georgia Institute of Technology
Atlanta, 1959-63

SUMMARY REPORT, PHASE I

PROJECT NO. A-425

PRELIMINARY DESIGN OF A HUMAN ANGULAR ACCELERATOR

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ORDER NO. K-6898C-59

U. S. ARMY MEDICAL RESEARCH LABORATORY

FORT KNOX, KENTUCKY

APRIL 1959

Engineering Experiment Station
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I. SUMMARY

This report concerns the preliminary design of a rotating device for studying the effects of angular accelerations with centrifugal force on human or animal subjects. The study includes the rotating beam, supporting base, and subject's chair. The prime mover and gear reduction are not within the scope of the present study. The purpose of the study is to establish the general configuration and power requirements.

II. DESCRIPTION AND ACCELERATOR

A. General Features

The device will consist of a horizontal beam mounted on a central vertical spindle. A chair will be mounted on the beam so that a subject may be positioned anywhere from the axis of rotation to the beam end. A servo-controlled electrohydraulic drive system and programmer, to be supplied by the sponsor, will permit rotation of the beam and chair at predetermined constant angular accelerations up to the terminal velocity. Constant deceleration from maximum terminal velocity through zero to maximum terminal velocity in the other direction is one of the desired objectives.

The present design includes a simple chair. The ultimate objectives for the accelerator include a complex chair capable of remotely programmed positioning of tilt, heading, and radius in any combination. The present design will be based on the structural loads imposed by the complex chair at maximum velocity and acceleration. Provisions will be made in the structure to permit future installation of the power operated chair with a minimum amount of modification.

1. Speed Range

The accelerator shall be capable of rotating in either direction at speeds from 0 to 60 rpm.

2. Acceleration Range

The accelerator shall be capable of angular accelerations from 0 to 2 rad/sec^2 with the maximum design load at the maximum design radial distance.

3. Maximum Design Load

The maximum design load will consist of a gimbal mounted chair positioned at a nominal radius of 4-1/2 feet, supporting a 250 pound subject from a pivot 2 inches above the head at an angular acceleration of 2 rad/sec^2 and a terminal velocity of 60 rpm.

4. Dimensions

The overall diameter of the device will not exceed 11 feet. An additional cleared radius of approximately three feet will be required around the periphery to permit the chair to clear the wall when the chair is at the maximum radial position and freely suspended.

B. Mechanical Features

1. Rotating Beam

The rotating beam will be a lattice framework fabricated from 6061-T6 aluminum tubing. The working stress will be based on the yield strength and a safety factor of 5. The vertical deflection at the end of the beam will not exceed 0.050 inch at maximum load and acceleration. The beam will be removable from the spindle to permit modification and balancing. The chair track width will be three feet. The beam depth will be kept to a minimum to be determined during the detailed design. Shelves 18 inches wide by 4 feet long will be attached to the beam to provide standing room and mounting surfaces for oscillographs, amplifiers, and miscellaneous equipment. Each shelf will be capable of supporting a static load of 500 pounds.

The chair support carriage will be capable of supporting the complex chair and subject at maximum acceleration and velocity simultaneously. The chair support rails will consist of two type C-1060 steel shafts, hardened to Rockwell C-58-63 for a minimum depth of 0.10 inch and centerless ground. The rail surface finish shall not exceed 30 micro-inches RMS and shall be straight within 0.001 inch per foot. The chair support carriage will be attached to the rails by means of ball bushings; the size to be determined during the detailed design. The ball bushings will be adjusted to obtain zero clearance between the carriage support and rail. The chair support will be positioned radially by means of an irreversible ball bearing lead screw. Backlash will be eliminated by preloading the screw, and travel stops will be provided at the beam center and end. A hand crank will be provided to preset the radial position of the chair support carriage. Provisions will be included for the future addition of a remotely operated servomotor to perform this function. Identical chair support carriages will be provided at each end of the beam. The "dead" carriage shall provide a means for attaching counterweights, instruments, and a "simple" chair. The subject's chair will have sufficient radial movement to permit location of a seated upright subject's head at the center of rotation.

Space will be provided at the center and ends of the beam for mounting power outlet junction boxes. An arch fabricated of 6061-T6 aluminum alloy will be attached to the auxiliary shelves to permit mounting of the low level slip ring box assembly over the center of rotation.

2. Chair

The chair referred to herein is the "simple" chair. The chair will consist of a framework of 6061-T6 aluminum tubing, beams, and stiffened aluminum panels. All strength calculations will be based on the yield strength and a safety factor of 5. The chair will be supported on the carriage by means of a large four point contact bearing to permit presetting of the chair heading. A clamp will be provided to lock the chair at any preset heading from 0 to 360 degrees. A dial and index will be provided on the chair base to permit reading heading angle to 1/2 degree or better. A scale and index will be attached to the beam and carriage to permit radial positioning to 1/32 inch. The chair and back will be padded for comfort. The chair will provide head and back support for a maximum distance of 42 inches above the seat. Approximately 5 inches vertical seat adjustment will be provided to permit location of the subject's head at the desired height. An additional 12 inches of height will be provided above the 42 inch level for support of instruments and light equipment. Restraint will be provided in the form of straps or clamps to secure the subject's trunk and limbs against movement under conditions of maximum velocity and acceleration. A safety factor of 5 based on the ultimate strength will be used in the selection of webbing connections and clamps. Encapsulating panels will be provided to prevent air movement over the subject during rotation. Attachment points will be provided on the chair arms for signaling devices and other light equipment.

3. Spindle and Bearings

The spindle will be machined from Type C-1045 steel, or equivalent. A flange will be supplied at the upper end to provide for the beam attachment.

A spline will be machined on the lower end to permit installation of a coupling. The spindle will be hardened to Rockwell C45-50, and ground all over. All threads will be ground. All shaft bearing seats will be bround to a surface finish of 50 micro-inches RMS maximum. The spindle will be supported and positioned with angular contact or tapered roller bearings. Means will be provided for adjusting the bearings to remove all radial and axial play. Bearings will be special-precision type (ABEC-3) and sound level tested prior to delivery to assure minimum noise level. The spindle and bearings will be assembled in a steel sub-housing or sleeve. The housing bearing seats will be ground to a surface finish of 50 micro-inches RMS maximum. The sub-housing will be screwed or bolted into the machine frame. The bearings will be equipped with non-contacting labyrinth type seals. The housing design will provide a means for relubrication, if relubrication is required, as well as a means for slushing out the old lubricant. The method of lubrication will be selected during the detailed design.

4. Machine Frame

The machine frame will consist of a weldment of structural steel and hot rolled steel plate. The frame will be enclosed with steel panels, removable for inspection and servicing. Three point support will be provided from machine to floor to permit precise leveling. A jig will be provided to permit use of the slip ring arch for leveling purposes. The machine supports will be fastened to the floor with 1/2 inch anchor bolts. All panels will be reinforced to minimize vibration. Space will be provided to permit installation of a transmission type to be selected during the detailed design, after the drive motor selection.

C. Electrical Features

1. Power System

The available power will consist of eight 110 volt A.C., 20 ampere circuits. The power leads will proceed from an external junction box or patch panel to the slip ring terminal board located on the slip ring brush block. Three junction boxes will be mounted on the rotating beam; one in the center, and one at each end. The conductors will be routed along the beam in an aluminum tubing conduit. The utility connectors will be Deutsch ball-lock, push-pull type. Five ground plugs will be provided along the "live" end of the beam, and 2 ground plugs on the "dead" end.

2. Power Slip Rings

The slip ring assembly will be mounted on the spindle. The sub-housing will be equipped with an access door to permit inspection and modification. The slip ring assembly will be of the pancake type with twenty 20 ampere rings. One ring will be connected to the beam grounding plugs and to an external ground. All utility plugs will be of the grounding type. The grounded contact will be connected to the ground slip ring.

3. Instrument Slip Rings

The instrument slip ring assembly will be mounted over the beam center on an aluminum arch. The slip ring box will be grounded and the low level conduit to the control console will be shielded. The low level slip ring assembly will be selected from production types, if possible. The rings will be coin gold, half hard temper, set in epoxy resin. The ring shaft and back shaft will be concentric within 0.003 inch. The individual leads will be shielded from the slip ring box to the beam.

4. Low Level Brushes

The low level brushes will be fabricated of Paliney No. 7 alloy, or equivalent, heat treated to 280 Brinell hardness, mounted in epoxy plastic. The brush contact pressure will not exceed 5 grams.

5. Acceleration-Limit Switch

One 20 ampere circuit or signal circuit will be connected to an adjustable acceleration limiting switch (0 $\pm$ 6 G) mounted at the beam end. The limit switch will be connected into the electric pump motor circuit or the servo valve circuit, to be determined. A motion-sensitive switch will be provided to permit installation of a safety door interlock to prevent entrance to the accelerator enclosure while the beam is moving.

D. Structure Analysis

1. Beam Design

A total weight of 500 pounds for man, complex chair and instruments is assumed to act at a distance of 5 feet from the center of rotation. The maximum angular acceleration is 2 rad/sec^2, and the maximum angular velocity is 2 π rad/sec. The maximum power requirement occurs at the moment when the maximum velocity and maximum acceleration exist simultaneously. The maximum forces due to "g" loading will continue at the maximum terminal velocity.

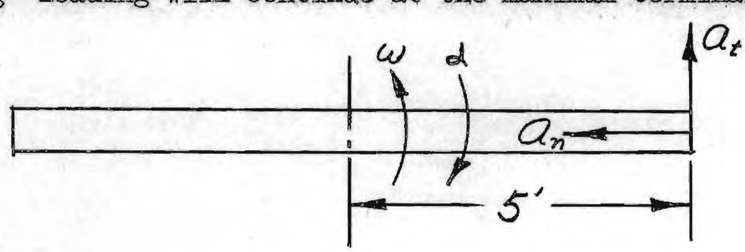


Figure 1. Load Diagram (Top View)

$$A_t = r\alpha = (5)\ (2) = 10\ \text{ft/sec}^2 = 0.31g$$

$$A_n = rw^2 = (5)\ (2\quad)^2 = 197.2\ \text{ft/sec}^2 = 6.12g$$

Centrifugal force, $F_c = MA_n = (\frac{500}{32.2})(197.2) = 3062$ lb.

Tangential force, $F_t = MA_t = MA_t = (\frac{500}{32.2})(10) = 155$ lb.

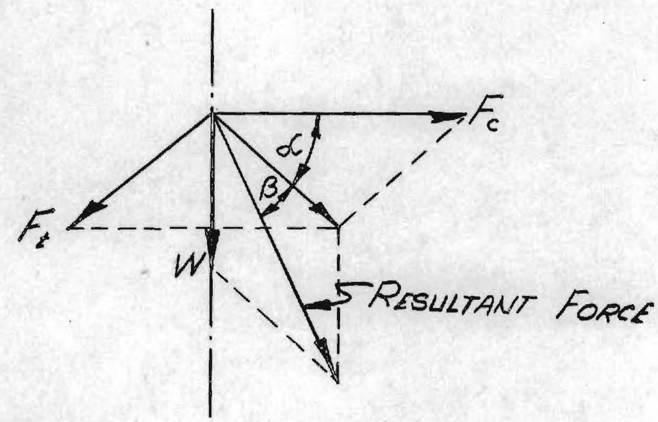


Figure 2. Force Diagram

$$\alpha = \tan^{-1}\frac{155}{3062} = 0.0506 = 3^{\circ}$$

$$\beta = \tan^{-1}\frac{500}{\sqrt{3062^2 + 155^2}} = \frac{500}{3065} = 0.1631 = 9\text{-}1/2^{\circ}$$

The force will be centered at a point about 60" above the top rail of the beam. The tangential component is neglected temporarily, and the stresses due to centrifugal and weight forces are considered.

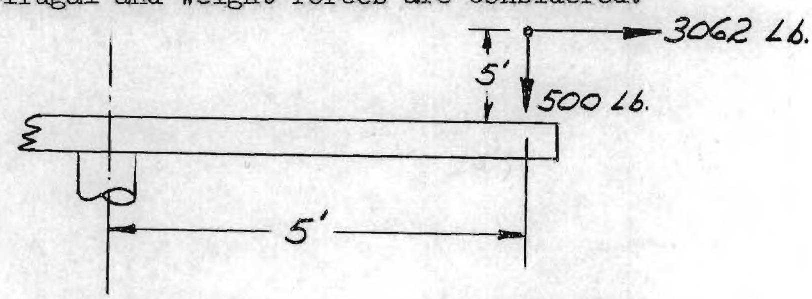


Figure 3. Load Diagram (Side View)

The above loading is equivalent to the following:

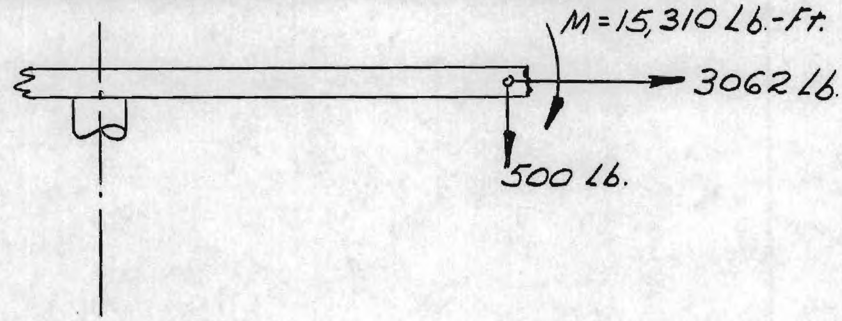
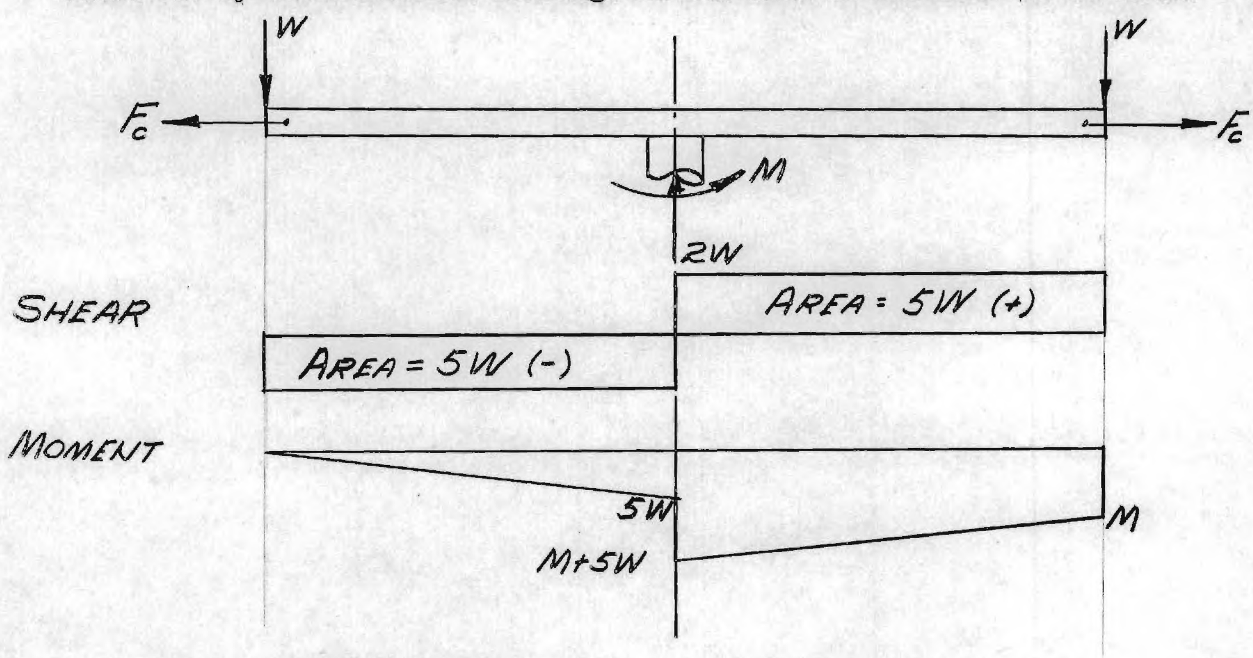


Figure 4. Equivalent Load Diagram

Now suppose a counterweight is mounted on the beam opposite the chair. The centrifugal force and the weight will be balanced, but the moment must be taken out by the shaft and bearings at the axis of rotation as follows:



Maximum moment = M + 5W = 15,310 + 5(500) = 18,810

Or approximately 19,000 lb-ft.

Figure 5. Shear and Moment Diagram

The beam will be constructed of 6061-T6 aluminum tubing. The working stress is based on the yield strength and a safety factor of 5. Yield strength

$= 37{,}000$ psi $\therefore$ $\sigma_{allowable} = \frac{37{,}000}{5} = 7400$ psi. The required moment of inertia of the beam at the center may now be calculated. Two requirements must be met: (1) the allowable tensile stress is 7400 psi. (2) The maximum deflection at the end should not exceed 0.050 inch.

The beam weight is neglected in the first calculation of the required moment of inertia. The following loading is assumed for the first estimate:

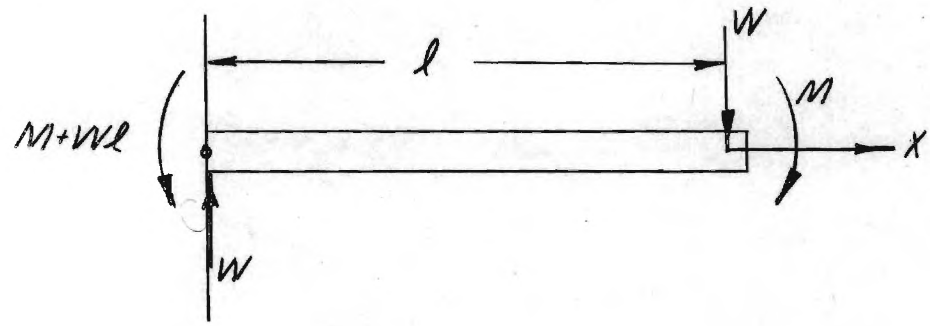


Figure 6. Beam Reaction Diagram

$W = 500$ lb.

$l = 5$ Ft. $= 60$ in.

$M = (19{,}000)(12)$(lb-in)

The maximum deflection occurs at $x = l$.

$$y_{max} = \frac{1}{3}\frac{Wl^3}{EI} + \frac{1}{2}\frac{Ml^2}{EI} \quad \frac{1}{EI}\left(\frac{Wl^3}{3} \quad \frac{Ml^2}{2}\right)$$

$$I = \frac{1}{y_{max}E}\left(\frac{Wl^3}{3} + \frac{Ml^2}{2}\right) = \frac{1}{(.050)\ (10^7)}\left[\frac{(500)(60^3)}{3} \quad \frac{(19{,}000)(12)(60)^2}{2}\right]$$

$$I_{required} = 893 \text{ in}^4$$

Calculating the required moment of inertia based on the allowable stress.

$$I_{req.} = \frac{Mc}{\sigma_{all.}} \quad \frac{(228{,}000)(6 \text{ in.})}{7400} = 185 \text{ in.}^4$$

where M = maximum bending moment

C = distance from neutral axis to extreme fiber.

The design must be based on the 893 in^4 moment of inertia required for minimum deflection. The bending moment used thus far is the maximum in the beam and is the sum of a constant moment and a linearly varying weight moment. A check will now be made to determine if it is practical to vary the beam depth.

The beam moment of inertia must not fall below the value required by the constant moment; therefore, that value will be used as a minimum.

Constant moment = (19,000)(12) in-lb.

$$I_{req'd.} = \frac{Ml^2}{2\ E\ y_{max}} = \frac{(19{,}000)(12)(60)^2}{2 \times 10^7 \times 0.050} = 821 \text{ in}^4$$

Suppose the section depth is decreased from 12 in. to 10 in. The moment of inertia becomes

$$I = 4\left[3.255 + 25(6.25)\right] = 638 \text{ in}^4$$

It will not be practical to taper the beam.

Numerous beam configurations were investigated in order to secure the required stiffness with minimum weight. The final beam configuration is shown below in cross-section

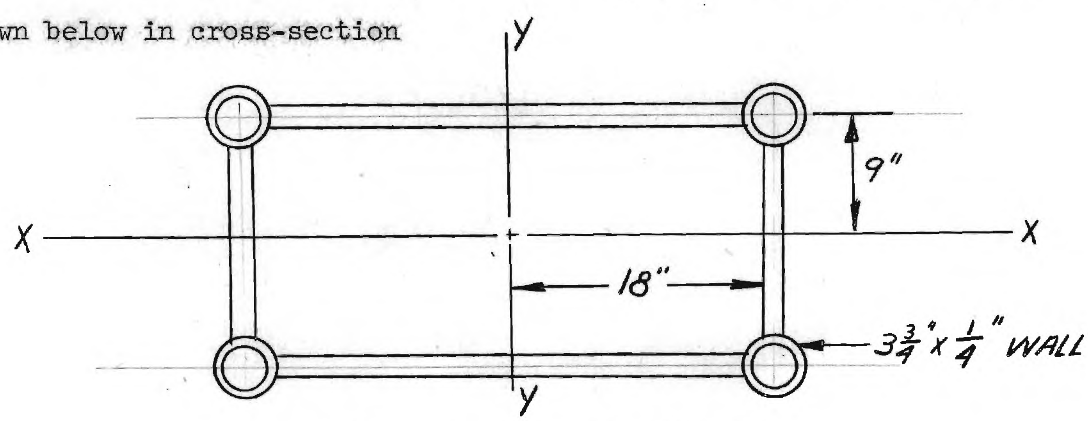


Figure 7. Beam Cross-Section

$I_p = 4.231 \text{ in}^4$

$A = 2.749 \text{ in}^2$

$$I_{yy} = 4\left[I_p + Ad^2\right] = 4\left[4.231 + 2.749(18)^2\right] = 3580 \text{ in}^4$$

$$I_{xx} = 4\left[4.231 + 2.749(9)^2\right] = 904 \text{ in}^4$$

2. Beam Intermediate Struts with Centrifugal Loading

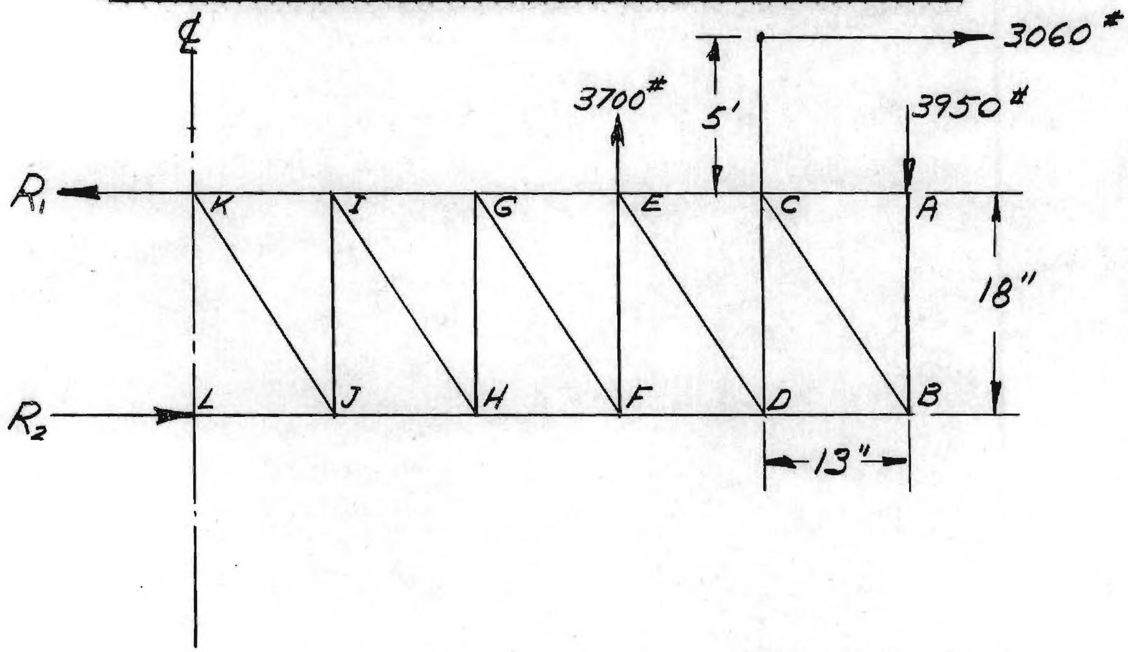


Figure 8. Intermediate Struts (Side View)

A joint by joint analysis was made of the slant bracing and vertical side members. The results are summarized below.

Member	Load	Direction
AB	3950	Compression
BC	4871	Tension
BC	2854	Compression

(continued)

Member	Load	Direction
DE	4871	Compression
DF	5708	Compression
EF	250	Compression
EG	5708	Tension
FH	5888	Compression
GH	250	Compression
GI	5888	Tension
HJ	6068	Compression
HI	308	Tension
IJ	250	Compression
IK	6068	Tension
JK	308	Tension
JL	6248	Compression
KL	250	Compression
R_1	6248	Tension
R_2	6248	Compression

The sizing of these members follows.

Maximum tensile force in side bracing = 4871 lb.

Maximum compressive force = 3950 lb.

Using 2" x 1/8" wall aluminum tube for side members:

$$\sigma_{actual} = \frac{4871}{01736} = 6618 \text{ psi. (tensile)}$$

$$\sigma_{allowable} = 7400 \text{ psi}$$

Required I for compression member using L/D = 60 = 0.003 in^4.

I_{actual} =).325 in^4

$\sigma_{actual} = \frac{3950}{01736} = 5367$ psi (compression)

3. Rail Forces

The chair load will be taken into the beam through a pair of circular rails. The analysis is summarized as follows:

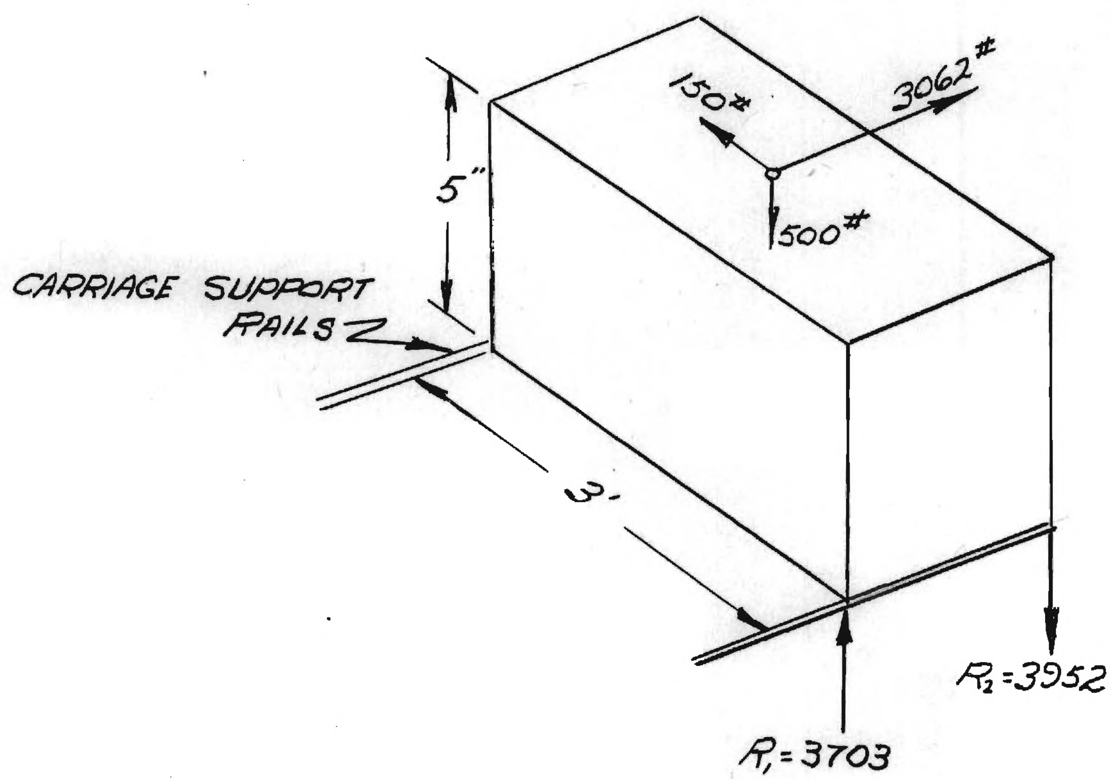


Figure 9. Chair Support Carriage Loading Diagram

$\sum M_{R_1}$ = 5(3062) - 2 R_2 + 500; R_2 = 7905 lb.

$\sum M_{R_2}$ = 500 - 7655 + $2R_1$ = 0; R_1 = 7405 lb.

The minimum rail size is limited by the allowable load on the ball bushings supporting the carriage. Five 2-1/2 inch diameter ball bushings are required on each side of the carriage as follows

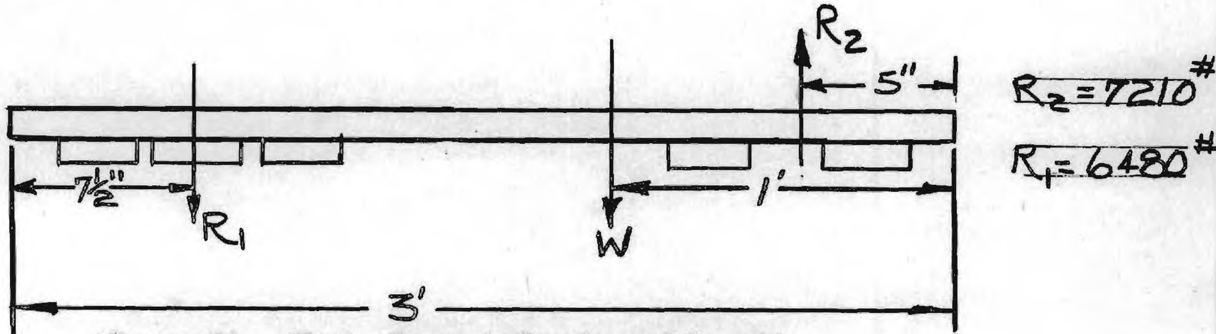


Figure 10. Chair Support Carriage Schematic

The carriage length is increased to 3 feet. This will permit movement of the subject's C.G. to a radius of 4.5 feet, however, the carriage will project 2 feet over center when the subject is centered. This will probably necessitate locating the counterweight inside the beam.

Using a bolt spacing of 6 inches and a maximum rail deflection of 0.010 inch, the required rail moment of inertia is as follows:

$$I = \frac{1}{48} \frac{Wl^3}{Ey_{max}} = 0.0469 \text{ in}^4$$

The 2-1/2 inch O.D. x 1/8 inch wall will satisfy the requirement.

4. Moment of Inertia and Power Requirement

The calculated moments about the spindle are as follows:

Subject and chair $I_{Z_M} = I_M + M\bar{r}^2 = 15 + (\frac{500}{32.2})\,(20.25) = 329.48$ lb-ft-sec^2

Counterweight $I_{Z_{cw}} = I_{cw} + M\bar{r}^2 = 13 + (\frac{500}{32.2})\,(20.25) = 327.48$ lb-ft-sec^2

Truss $I_Z = \frac{Ml^2}{12} = \frac{335}{32.2} \cdot \frac{(121)}{12} = 105.04$ lb-ft-sec^2

Rails $I_Z = \frac{Ml^2}{12} = \frac{231(121)}{32.2(12)} = 72.34$ lb-ft-sec^2

Shaft (composite) $I_Z = 1.813$ lb-ft-sec^2

Coupling $I_Z = 0.286$ lb-ft-sec^2

Total Moment of inertia to the transmission = 836.59 lb-ft-sec^2

Torque to the transmission = I = (836.59)(2) = 1673.18 lb-ft.

Required horsepower = $\frac{TN}{5252}$ = 19.11

Moment of inertia with subject centered = 207.59 lb-ft-sec^2

Torque required = (207.59)(2) = 415.18 lb-ft.

Required horsepower = $\frac{(415.18)(60)}{5252}$ = 4.75

5. General Remarks

All dimensions, weights, and forces are estimated. The exact values will be determined during the detailed design. The final design will be determined by the degrees of freedom and range of movement of the "complex" chair. The maximum force values will be close to those calculated for the case where the subject's chair is suspended from a pivot 2 inches above the subject's head and freely suspended. This configuration introduces a large moment due to the height of the pivot above the beam. The estimated beam weight for this maximum condition is 200 pounds. It should be noted that the beam moment of inertia is not significant compared to the moment of inertia of subject and chair at the maximum radial position.

The tachometer and heading indicators will be selected during the detailed design.

Respectfully submitted

Winston C. Boteler
Project Director

Approved:

Thomas W. Jackson, Chief
Mechanical Sciences Division

APPENDIX I.

Figures

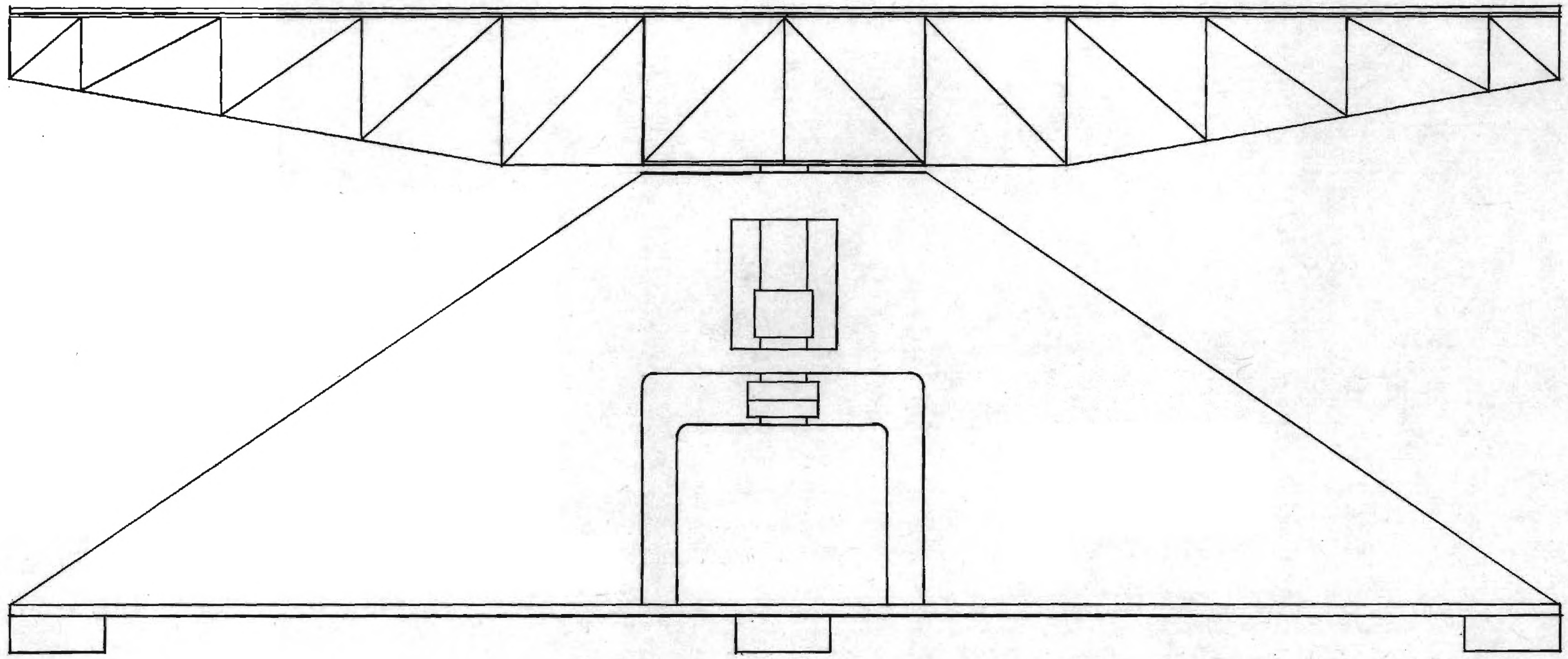
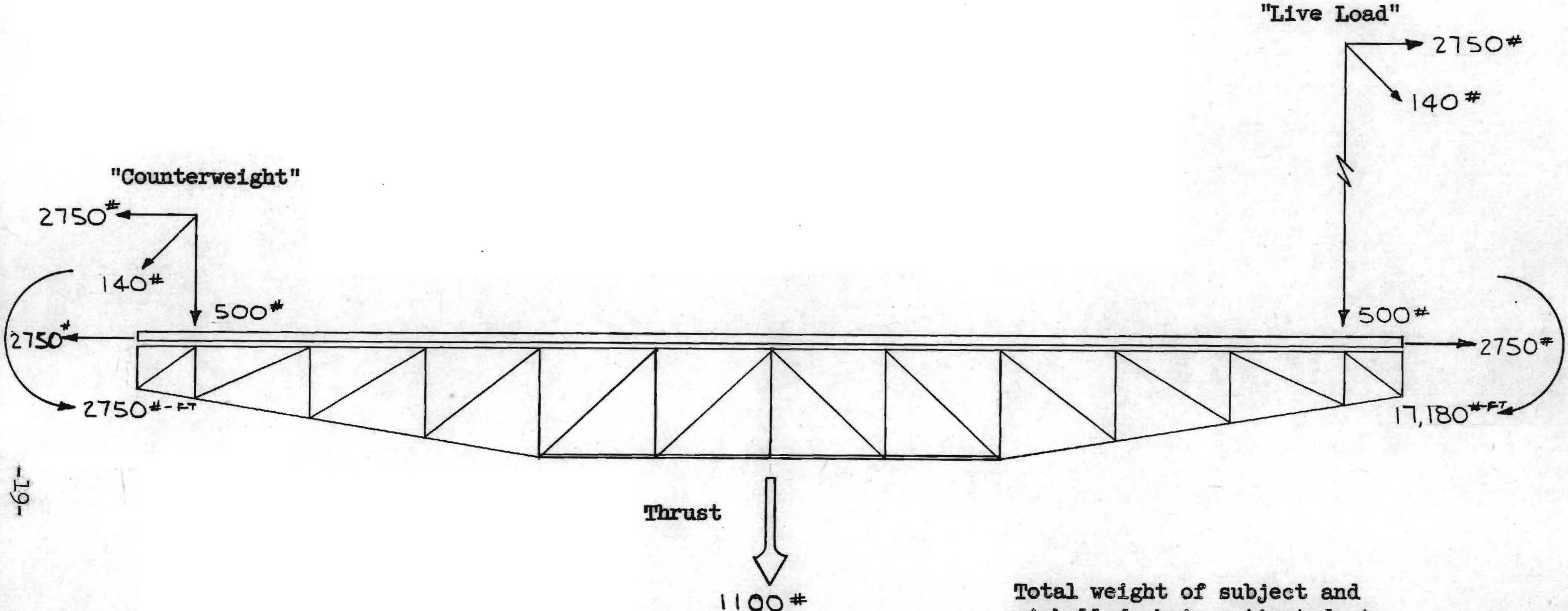


Figure No. 11 Rotating Beam & Support

Project No. A-425



Total weight of subject and gimballed chair estimated at 500 lb.

Figure No. 12 Load Diagram - Rotating Beam

Project No. A-424

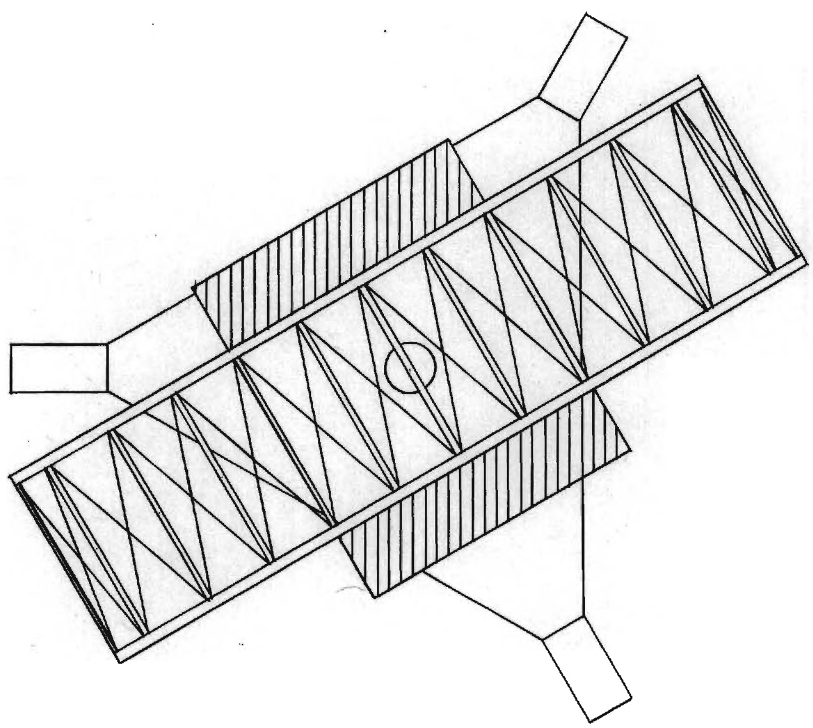


Figure No. 13 Rotating Beam and Support, Top View

Project No. A-425

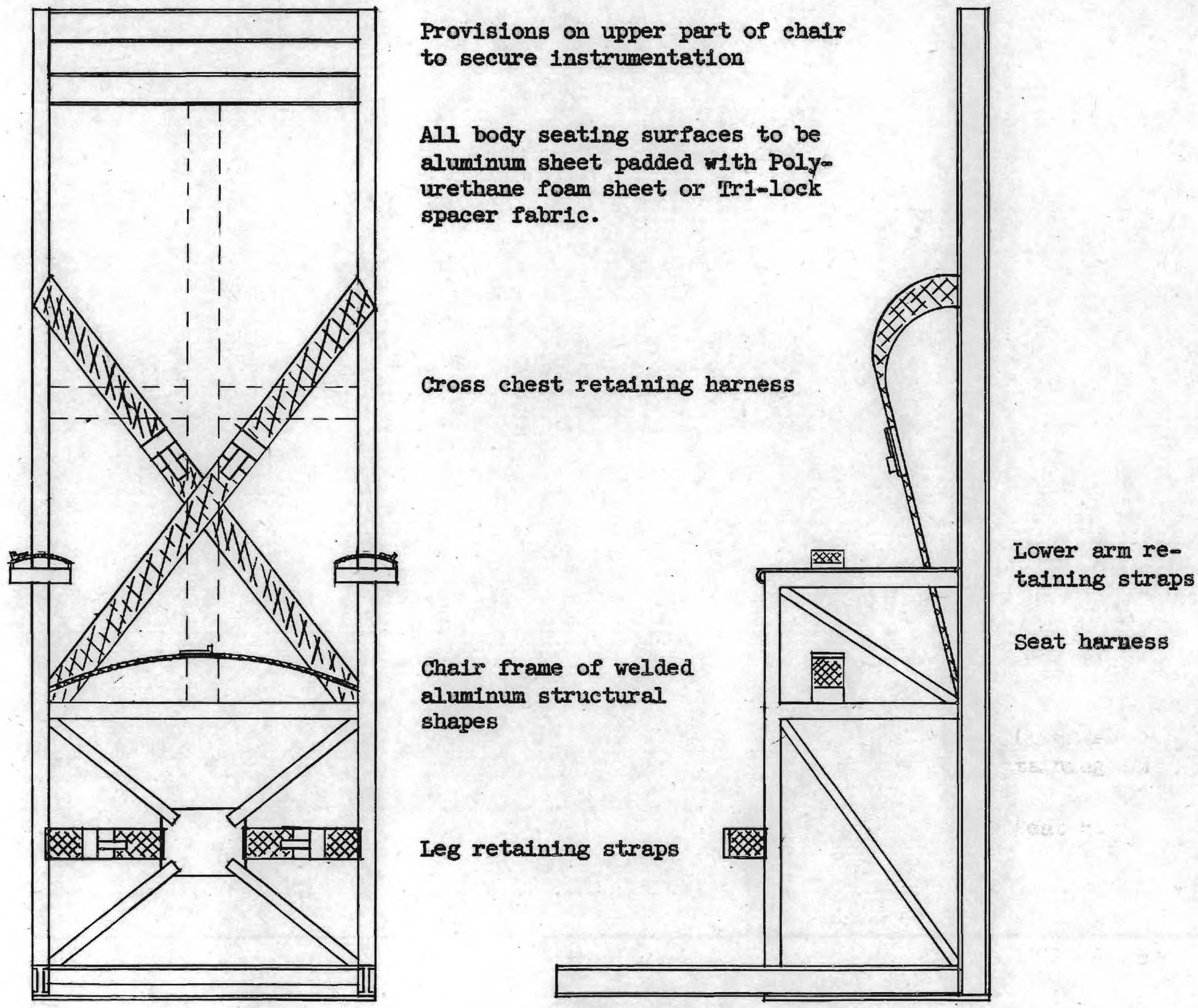


Figure No. 14 Schematic Drawing for Simple Chair
Project No. A-425

Ga Tech
A425/FR/1063

FINAL REPORT

PROJECT A-425

RESEARCH AND DEVELOPMENT OF A HUMAN ANGULAR ACCELERATOR

WINSTON C. BOTELER

Contract No. DA-16-015-AII-6729

October 1963

Prepared for
U. S. Army Medical Center
Research Laboratory
Fort Knox, Kentucky

1963

Engineering Experiment Station
GEORGIA INSTITUTE OF TECHNOLOGY
Atlanta, Georgia

ENGINEERING EXPERIMENT STATION
of the Georgia Institute of Technology
Atlanta, Georgia

FINAL REPORT

PROJECT NO. A-425

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CONTRACT NO. DA-15-014-AII-6729

6

OCTOBER 1963

Performed for
U. S. ARMY MEDICAL CENTER
RESEARCH LABORATORY, FORT KNOX, KENTUCKY

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FOREWORD

This report contains a summary of the development of a rotating device for use in studying the vestibular response of human subjects to angular acceleration.

The project was sponsored by the U. S. Army Medical Center, Fort Knox, Kentucky, under the technical cognizance of Major George Crampton.

ACKNOWLEDGEMENTS

The author wishes to express his appreciation to the following people who were responsible for the detail design and fabrication of the system: Lee H. Knight, Frank H. Smith, William L. Tucker, and J. J. Foust.

ABSTRACT

A device consisting of a rotating beam and chair for vestibular stimulation of human subjects is described. The rotating assembly may be programmed for constant angular accelerations ranging from 0.05 deg/sec^2 to 400 deg/sec^2 and sinusoidal programs up to 1/4 cycles/sec with maximum velocity of 50 revolutions per minute.

The chair may be moved from the center of rotation to a radius of 4-1/2 feet while the beam is rotating. A capsule on the chair permits isolation of the subject from light and sound cues.

An electro hydraulic servo system provides the motive power.

The system includes power and slip rings for transmission of signals to and from the rotating assembly.

I. INTRODUCTION

The growing interest in the effects of space flight on pilots has increased the need for a better understanding of the vestibular mechanism in the inner ear.

Dynamic studies of the vestibular organ are difficult. Since the canals respond only to angular acceleration, any laboratory experiment must permit the application of a controlled angular acceleration to the subject for canal stimulation.

Ideally, the stimulator would rotate the subject at a constant angular acceleration up to a preselected terminal velocity without noise or vibration. The angular accelerator described provides a means for providing controlled angular acceleration, as well as linear acceleration of varying magnitude.

II. GENERAL DESCRIPTION OF THE APPARATUS

The human angular accelerator consists of two major components; the rotating beam and chair assembly and the electrohydraulic servo system. Figure 1 shows the entire assembly during installation. The rotating system consists of a horizontal beam 11 feet long mounted on a vertical spindle in a fabricated steel pedestal. The spindle is driven by a rotary hydraulic actuator through a 55:1 ratio precision helical gear reducer. The hydraulic motor speed is controlled by a four way servo valve manifolded to the motor, which is controlled in turn by a two stage integrating servo amplifier. The command signal is supplied by a D.C. function generator. Velocity feedback to the control system is provided by a D.C. tachometer driven directly from the hydraulic motor.

Power and communication circuits are supplied to the chair during rotation by a set of 26 slip rings mounted on the spindle shaft. A set of 30

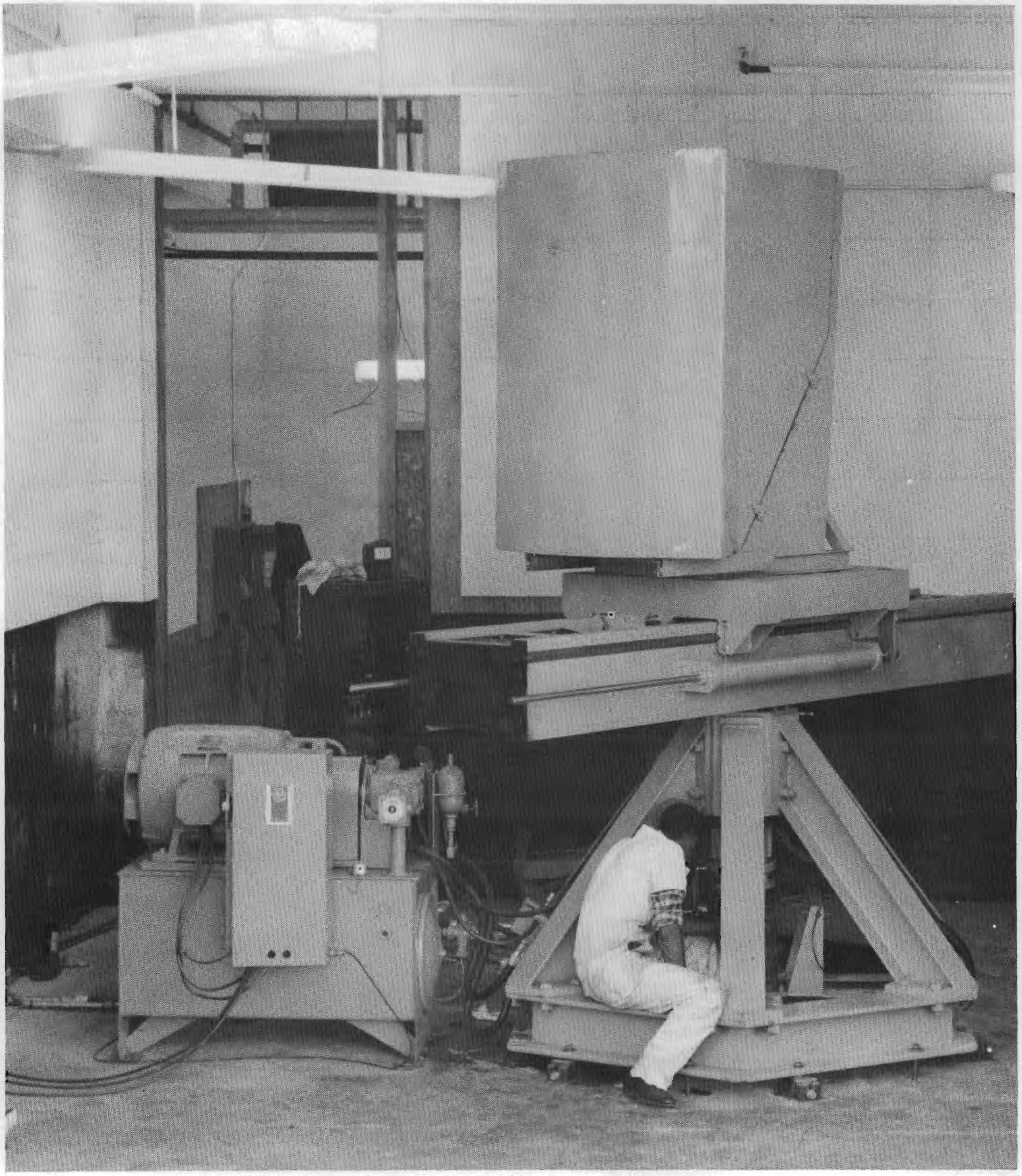

Figure 1. Angular Accelerator Assembly.

slip rings mounted on the ceiling provides for the transmission of low level signals from electrodes and preamplifiers.

III. ROTATING BEAM AND PEDESTAL

A. Beam, Chair and Canopy

The rotating beam consists primarily of two 12 inch aluminum channels, 11 feet long, bolted up 22 inches apart. The working stress was based on the yield strength and a safety factor of 5. The maximum design load was dictated by the future requirement of a gimbal mounted chair positioned at a nominal radius of 4-1/2 feet, supporting a 250 pound subject from a point 2 inches above the head, at an angular acceleration of 1 rad/sec^2 and a terminal velocity of 50 revolutions per minute. The beam is bolted to the spindle through a welded box beam.

The chair support carriage is of welded aluminum construction. The chair load is transmitted to the beam by four recirculating roller bearings, which run on hardened steel vee ways bolted to the sides of the beam. A motor driven ball screw moves the carriage along the beam. A 24 inch diameter 4 point contact ball bearing mounted on the carriage supports the chair. A manually controlled double worm and gear set permits 360° positioning of the chair on the carriage.

The chair is of welded aluminum tubing construction. The seat, back, and arms are fiberglas covered to insulate the subject. Seat, back, and arm cushion of 1/2 inch thick Trilok fabric further isolate the subject from the chair. Lap and leg belts and a shoulder harness provide restraint for the subject under conditions of maximum velocity and acceleration.

A sliding canopy attached to the chair encloses the subject and permits full control of ambient lighting.

B. Pedestal

The pedestal consists of a weldment of structural steel "I" beams. The spindle shaft is positioned by two precision tapered roller bearings. A multiple disc type coupling at the lower end of the shaft connects the spindle to the gear reducer output shaft. The gear reducer is a 55:1 ratio three stage combination bevel and helical gear set rated for a maximum torque of 12,500 in-lb. All gears are ground or lapped for a maximum backlash of 0.002"-0.003". The gear reducer input is coupled to the hydraulic motor with a multiple disc type coupling.

IV. ELECTROHYDRAULIC SERVO SYSTEM

A. General Description

The electro hydraulic servo system consists of three major components; hydraulic power supply, servo amplifier, and hydraulic actuator. The servo system schematic diagram is shown on Figure 2.

B. Servo Components

1. Servo Amplifier

The velocity control system is a zero error (Type 1) servo system which uses an integrating amplifier. The output of the Vickers Model EAPS-D-11 integrating amplifier is proportional to the time integral of the input instead of being simply proportional to the input.

This is accomplished by means of a capacitive negative feedback loop within the amplifier. The integrating amplifier gain becomes a rate of change of output for a given input; i.e., milliamperes per second per volt input instead of simply milliamperes per volt input. In this system the amplifier has

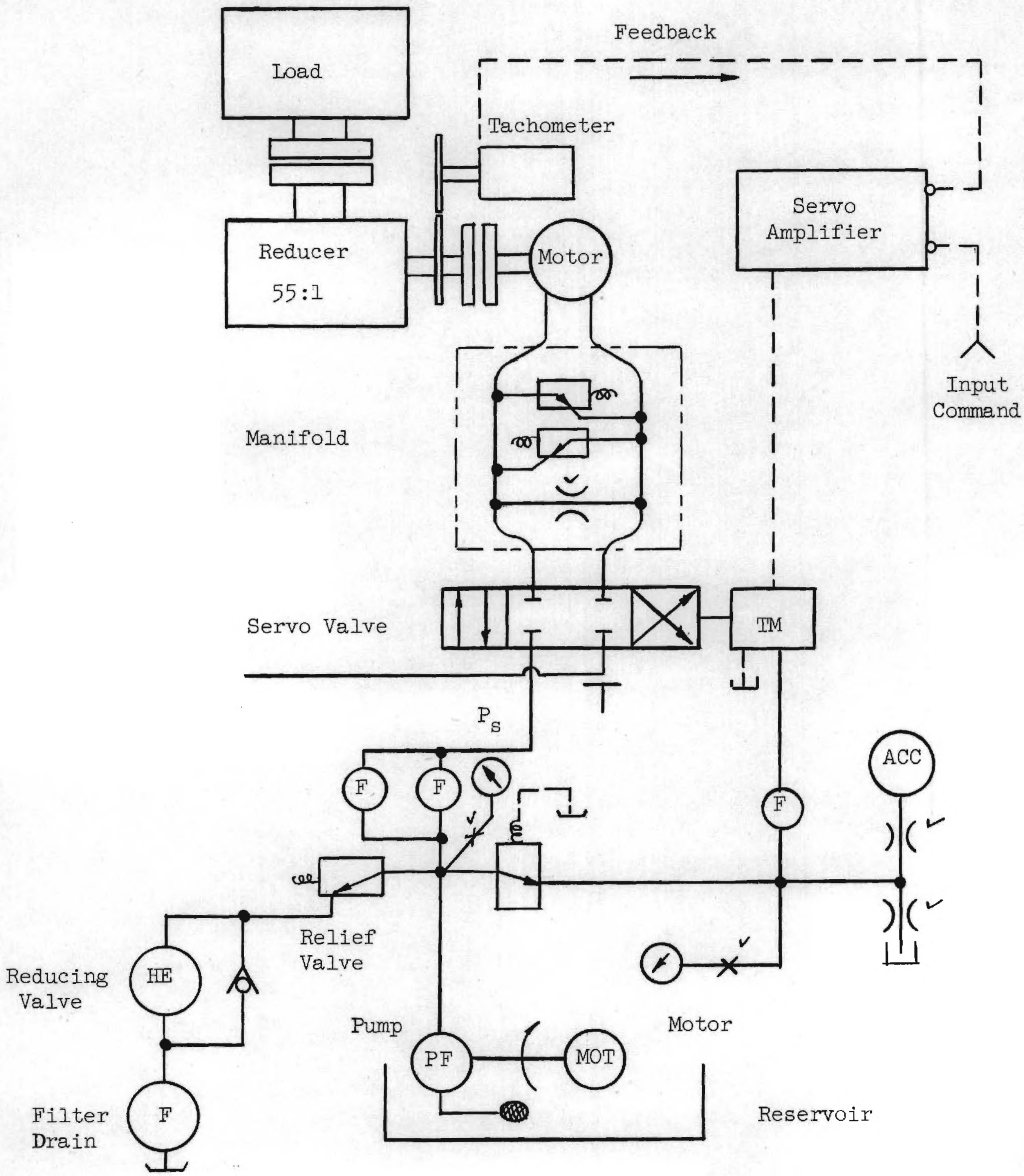

Figure 2. Electrohydraulic Servo System Schematic

two inputs. The D.C. command function generator output and the D.C. tachometer feedback signal are summed through resistors to provide the amplifier input signal.

2. Servo Valve

The servo valve is a Vickers Type SA4-06-40 with a linkage ratio setting (ratio of main spool to pilot spool movement) of 5.5 to 1. The maximum flow is 37 gallons per minute with the load pressure 66% of the supply pressure. The valve is manifolded directly to the servo motor. This minimizes the response time between the valve and motor.

3. Hydraulic Power Supply

The hydraulic power supply consists of a 100 gallon reservoir, vane type pump, 50 horsepower motor, and associated auxiliary equipment. A water to oil heat exchanger is included to permit continuous operation for long periods of time. The pump has a peak capacity of 42 gallons per minute at a pressure of 2000 pounds per square inch. Flexible hose connections are used between the power supply and the servo valve to minimize the transfer of vibrations to the rotating system. The power supply is mounted on vibration isolators to prevent transfer of vibrations through the floor to the pedestal.

4. Hydraulic Motor

The hydraulic actuator is a Vickers Type MF-2008-A-30-38-21 axial piston motor with a maximum output torque of 380 in-lbs at a speed of 2750 revolutions per minute (rpm). The minimum controllable speed is about 5 rpm. A D.C. tachometer with an output of 20 volts per 1000 rpm is geared 1:1 to the hydraulic motor with a set of anti-blacklash gears.

V. ELECTRICAL FEATURES

A. Power System

The power available to the rotating structure consists of eight 115 volt A.C., 20 ampere circuits. The power leads proceed from an external junction box or patch panel to the slip ring brush block. Three junction boxes are mounted on the rotating beam; one in the center, and one at each end. The conductors are routed along the beam in an aluminum tubing conduit.

B. Power Slip Rings

A 26 ring pancake type slip ring assembly is mounted on the spindle shaft extension between the lower bearing and the coupling flange. One ring is connected to the grounding plugs on the rotating beam.

C. Instrument Slip Ring

The instrument slip ring assembly consists of 30 coin-gold rings, half hard temper, set in epoxy resin. The brushes consist of .007 inch diameter Paliney alloy wires with a current capacity of 0.9 amperes. The ring and brush assembly is mounted in an aluminum housing which is attached to the ceiling above the center of rotation. A flexible cable connects the ring assembly to the electrode patch panel on the chair. Another cable connects the brush block to a patch panel in the control console.

VI. RESULTS

The angular accelerator was tested at Georgia Tech prior to shipment. Due to the inadequate floor mounting available, the tests were run with the chair at the center of rotation. A 250 pound dummy load was strapped in the chair for all tests. A Servomex Type LF51 low frequency function generator was used to provide the command signals. Sinusoidal velocity and constant angular acceleration programs were run. The constant angular acceleration or ramp program was accomplished by decelerating from maximum speed (50 rpm) in one direction and accelerating to maximum speed in the other direction. The device operated smoothly at accelerations varying from 0.05 deg/sec^2 to 200 deg/sec^2.

Sine wave response tests were run at frequencies ranging from 1/4 cycles/sec to 1/2000 cycles/sec with maximum velocities of 50 rpm in both directions. Response of the system was flat to 1/4 cycles/sec at maximum velocity amplitude. Runs were made with human subjects at constant angular accelerations up to 100 deg/sec^2 and sinusoidal frequencies to 1/6 cycles/sec. The transition through zero velocity was very smooth with no palpable transients.

After installation at the Army Medical Research Laboratory, Fort Knox, Kentucky, a sound insulated floor was erected which physically separated the rotating assembly from the hydraulic power supply. The hydraulic power supply was mounted on vibration isolators, thus eliminating almost all of the structure borne vibration. Some slight vibration was transmitted through the flexible hydraulic lines to the hydraulic motor, but this was not evident at the subject's chair. The floor effectively reduced and diffused the noise from the hydraulic power supply and eliminated this unit as a positional reference cue.

Tests were conducted at AMRL, Fort Knox, with the chair at the end of the beam and loaded with 288 pounds of dry cement.

The device operated smoothly at accelerations ranging from 0.1 deg/sec^2 to 60 deg/sec^2. Transition through zero velocity again was very smooth. Reversal was unnoticeable at the subject's chair.

A complete linearity and frequency response analysis will be performed when the complete programing system is installed.

Respectfully submitted:

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Project Director

APPENDIX I

OPERATING AND MAINTENANCE INSTRUCTIONS

OPERATING AND MAINTENANCE INSTRUCTIONS

A. Electrohydraulic Servo System

1. Hydraulic Power Supply

Details for connecting the hydraulic system are shown on Drawing No. A425-906. Prior to operation the reservoir should be checked to see that the fluid level is between the high and low level marks. Only high grade lubricating oil of the Hydraulic Oil Class with a viscosity rating of 300-SUS @ 100° F (MED) should be used in the system. Gulf Harmony "C", Sinclair Duro Oil 300, Shell Tellus 33, or equivalent oils are satisfactory. The system has been flushed and will not require additional flushing unless disassembled.

Should it become necessary to remove the hydraulic lines, extreme care should be taken to avoid getting dirt particles into the open ends. Clean pipe caps should be prepared for installation on the end fittings as they are removed. Any oil added to the system should be filtered through a 200 mesh screen. The reservoir should be drained and refilled every 12 months.

All bolts and machine screws on the hydraulic power supply should be checked for tightness once a week if the unit is operated regularly. The pipe fittings on both ends of the flexible actuator pressure lines should be checked frequently for tightness. The filter pressure gage on the hydraulic pump should be examined during each day's operation. When the filter pressure rises to the "REPLACE" section of the dial, the filter should be replaced.

The system should never be started under load. The by-pass valve should be opened before starting the motor. After the motor reaches operating speed the valve is closed. The system operating pressure should be set at 2,000 lb/in^2.

Water should be flowing through the oil cooler prior to starting the system. The flow should be adjusted during system operation to maintain the system temperature below 170° F.

The servo valve control pressure should be set at 1,000 lb/in^2.

2. Electrohydraulic Servo Valve

The servo valve requires no maintenance. Extreme care should be taken to keep dirt out of the manifold if the servo valve is removed for any reason. A clean blank plate should be installed over the manifold ports. It should be noted that the flushing procedure does not permit flushing of the manifold block and motor, therefore, openings in these parts must be covered immediately upon disassembly.

3. Servo Amplifier

The Vickers EAPS-D-11 Series servo amplifier is a two stage direct coupled amplifier, capable of amplifying D.C. input voltage. If the amplifier is not balanced an output signal will be present when the input is zero.

Electrical connections for the amplifier are shown on Drawing A-425-906. Prior to starting the hydraulic system, the amplifier should be checked for balance by noting the torque motor coil currents on the control console meters. These should read the same with zero input to the amplifier. Do not start the system until these currents are balanced, because the beam will rotate as the pressure rises. The amplifier balancing procedure is detailed in the amplifier manual.

4. Hydraulic Motor

The hydraulic motor requires no maintenance. The hydraulic connections should be checked frequently for tightness. The gears coupling the motor and tachometer should be checked frequently for proper mesh.

B. Gear Reducer

Prior to starting, the gear reducer should be filled to the indicated level with straight mineral oil, AGMA No. 4, viscosity range 700-1000 SUV @ 100° F. Typical oils are Gulf Parvis Oil H, Sinclair Duro No. 900, and Texas Regal G.

The unit should be given a daily routine inspection consisting of a visual inspection and observation for oil leaks or unusual noises. If either occur, the unit should be shut down at once, and the cause of the leakage or noise found and corrected. The oil level should be checked once a week. The maximum temperature of the oil in the case should not be allowed to exceed 180° F.

The oil should be changed after 2500 hours of operation or 6 months of normal operation, whichever occurs first.

Every precaution should be taken to prevent any foreign matter from entering the gear case. If it becomes necessary to shut down the unit for a period longer than one week, it will be necessary to run the unit for at least ten minutes each week it is idle. This operation will keep the gears and bearings coated with oil and prevent rusting due to condensation of moisture.

APPENDIX II

LIST OF DRAWINGS

LIST OF DRAWINGS

LIST OF DRAWINGS (Continued)

LIST OF DRAWINGS (Continued)